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CONDENSER AND FAN DEVELOPMENT FOR AUTOMOTIVE RANKINE CYCLE ENGINES

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Garrett Corporation

Prepared for:

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18 February 1972

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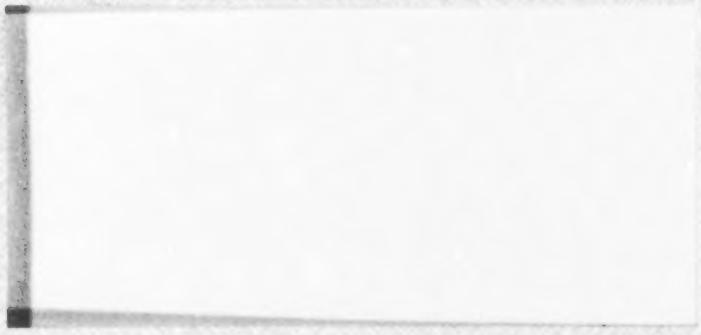

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QUARTERLY PROGRESS REPORT

CONDENSER AND FAN DEVELOPMENT FOR AUTOMOTIVE RANKINE CYCLE ENGINES

CONTRACT 68-01-0407
NOV. 10, 1971 TO JAN 31, 1972

72-8095(I)

February 18, 1972

Prepared for the
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Ann Arbor, Michigan 48105

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SECTION I

INTRODUCTION

Condenser size and cooling air power required for Rankine-cycle power plants have been a significant factor in limiting the applicability of this type of power plant for automobiles. The compact condenser development program recently completed by AiResearch resulted in a substantial condenser size reduction compared to that obtained using present day radiator technology. Other significant improvements are possible, especially with regard to an increase in the cooling air fan efficiency and the optimization of the condenser-fan installation.

The goal of the present study is to develop preprototype condenser, duct, and fan packages for each of the three Rankine cycle automotive power plants now under development. The study will involve the further optimization of the air-side perforated fin, the measurement of condensing coefficients, the optimization of the cooling system installation, and the design, fabrication, and testing of the fans and condenser.

SECTION 2

GENERAL PROGRESS

OVERALL

During the first quarterly reporting period, work was initiated in all seven tasks. The schedule was revised in several areas and is shown in its updated version in Figure 1. Reasons for the schedule changes and the progress in each task are summarized below.

TASK I -- FABRICATE AND TEST NEW SURFACES

During the previous program, perforated fins with 0.015-, 0.030-, and 0.060-in.-wide slots and open area ratios of 12.5 and 25.0 percent were tested, and the best overall performance was obtained for the 0.030-in. slots at 25.0 percent open area. On the basis of this work, it was decided to test the same slot width at open area ratios of 16.7 and 33.3 percent during the current program. The fin geometries are as follows:

Core	Slot Width, in.	Slot Length, in.	Slot Spacing, in.	Percent Open Area	Fin Height, in.	Fin Thickness, in.	Number Fins/in.
21	0.030	0.250	0.180	16.7	0.326	0.004	22
23	0.030	0.250	0.090	33.3	0.326	0.004	22

Figure 2 shows the perforation layout for these two fins.

It had been planned to select the third perforation geometry on the basis of a study of heat transfer from perforated plates at the University of Michigan (Contract No. 68-04-0019). During December a meeting was held with Dr. Wen-Jei Yang who had been conducting the study, and it was concluded that one of the two geometries already selected represented very nearly the optimum geometry (fin spacing and percent open area) indicated by the results of the University of Michigan study.

Therefore, it was decided to utilize the -13 fin geometry which is the present optimum, but to use a different fin height for the third test core. The fin height will be selected on the basis of condenser optimization studies which will be completed in February. In these studies, the fin geometry will be varied to establish the tradeoffs between fin geometry, condenser size, and condenser air horsepower.

The perforated fin stock for all three fins was received in January. Drawings were prepared and details were fabricated for the first two cores. These units will be brazed in February.

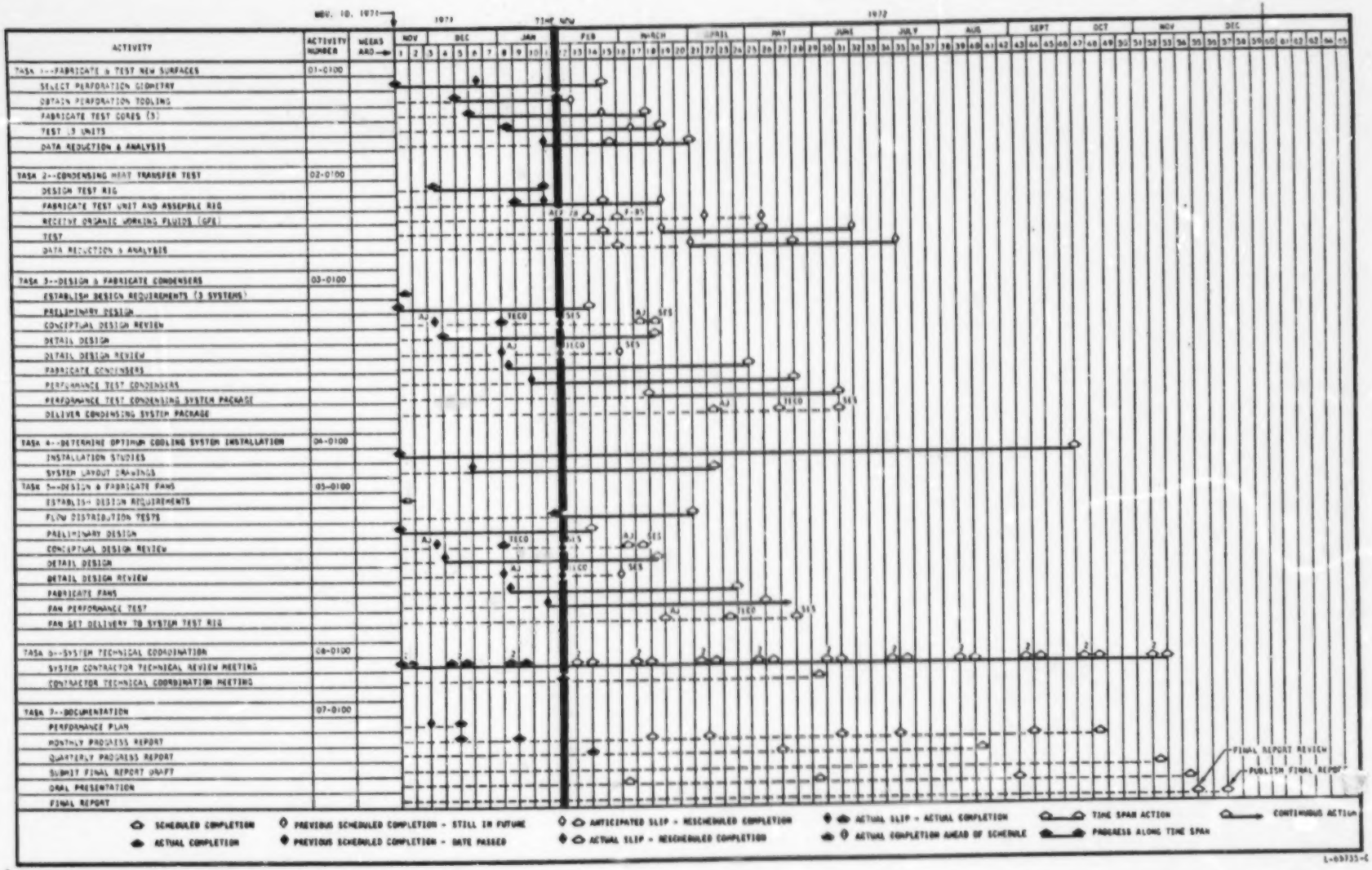
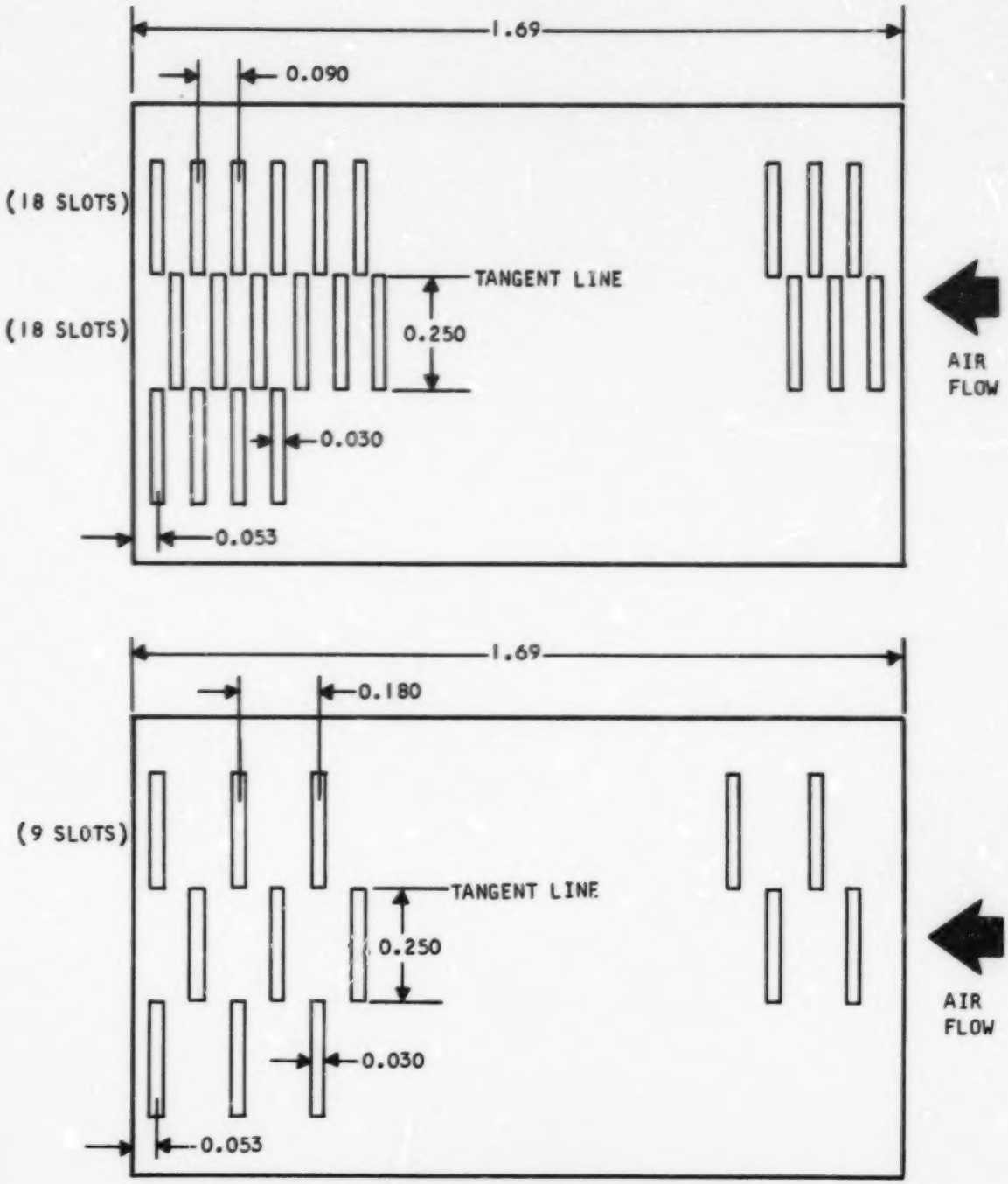


Figure 1. Program Schedule

RESEARCH MANUFACTURING COMPANY



NOTE: ALL DIMENSIONS ARE IN INCHES

S-68091

Figure 2. Perforated Fin Materials For Cores 21 and 23

A test plan was prepared and assembly of the setup shown schematically in Figure 3 was begun. The tests will be water-to-air tests with temperature, flows, and pressures recorded to obtain f and j data. The test conditions are as follows:

Air Flow Rate, lb/min	45	37.5	30	24	21	19.5	18	16	14	12.5	8.5
Water Flow Rate, lb/min	65	54	43.5	35	30.5	28.5	26	23	20.5	18.5	15
Air Inlet Temperature	LAB AMBIENT										
Water Inlet Temperature, ^{o}F	190										

TASK 2 -- CONDENSING HEAT TRANSFER TEST

During Task 2, condensing coefficients will be measured for the three working fluids. The condensing correlations of Soliman, et al (Reference 1) and Rohsenow, et al (Reference 2) have been modified to account for the friction factor of an offset fin, and have been used in the preliminary designs. These coefficients are shown as a function of vapor quality for the TECO design conditions in Figure 4. The condensing coefficient tests will establish the validity of these correlations by yielding coefficients at the design points of the condensers. It is planned to measure coefficients with small amounts of oil in the inlet vapor for the SES and TECO working fluids.

The condenser test unit is shown schematically in Figure 5. The unit is a cross-flow design with five parallel water passages as the heat sink. The vapor passage is 0.050 in. high by 3.4 in. wide by about 21 in. long. The fin in this passage is of rectangular offset geometry with 20 .004-in.-thick fins per inch. The water flow rate through each of the five sections is separately variable to provide control of the rate of change of quality along the vapor flow length. The heat load in each section is determined from measurements of water flowrate and inlet and outlet temperatures. Thermocouples in the vapor flow and in the walls of the vapor passage provide a wall-to-vapor ΔT which is used with the heat load to establish the condensing coefficients. Pressure taps in the condensing passage give pressure drop data as well as checks on the local saturation temperature of the vapor.

The condensing test facility is shown schematically in Figure 6. Plant steam is used to boil the liquid in the closed loop system. A glass sight tube and a glass condenser/subcooler are incorporated into the system to verify complete condensation through the test unit.

Design of the test unit has been completed and fabrication drawings have been prepared. Assembly of the test rig has begun with details of the oil separation apparatus still be be resolved.

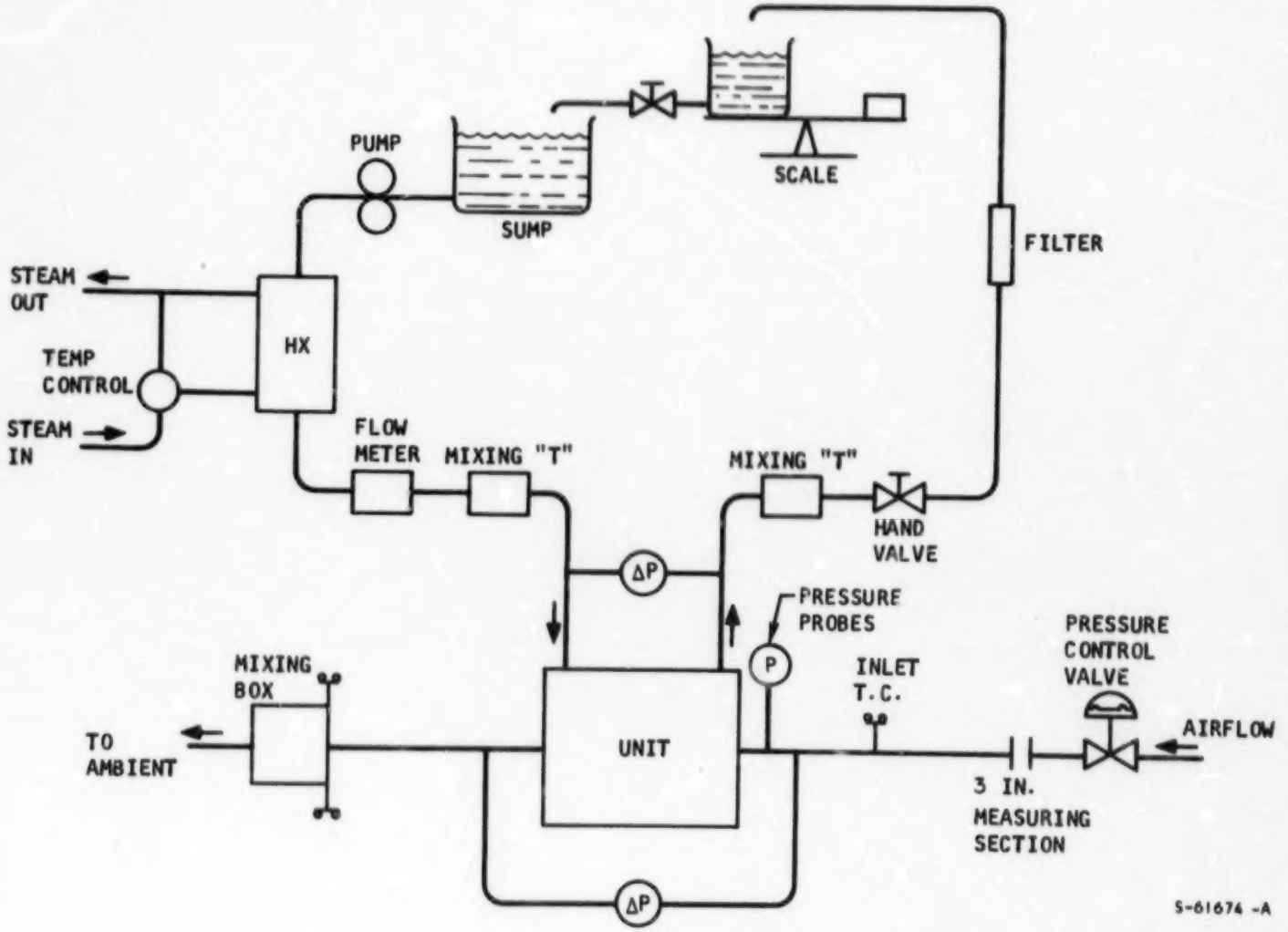
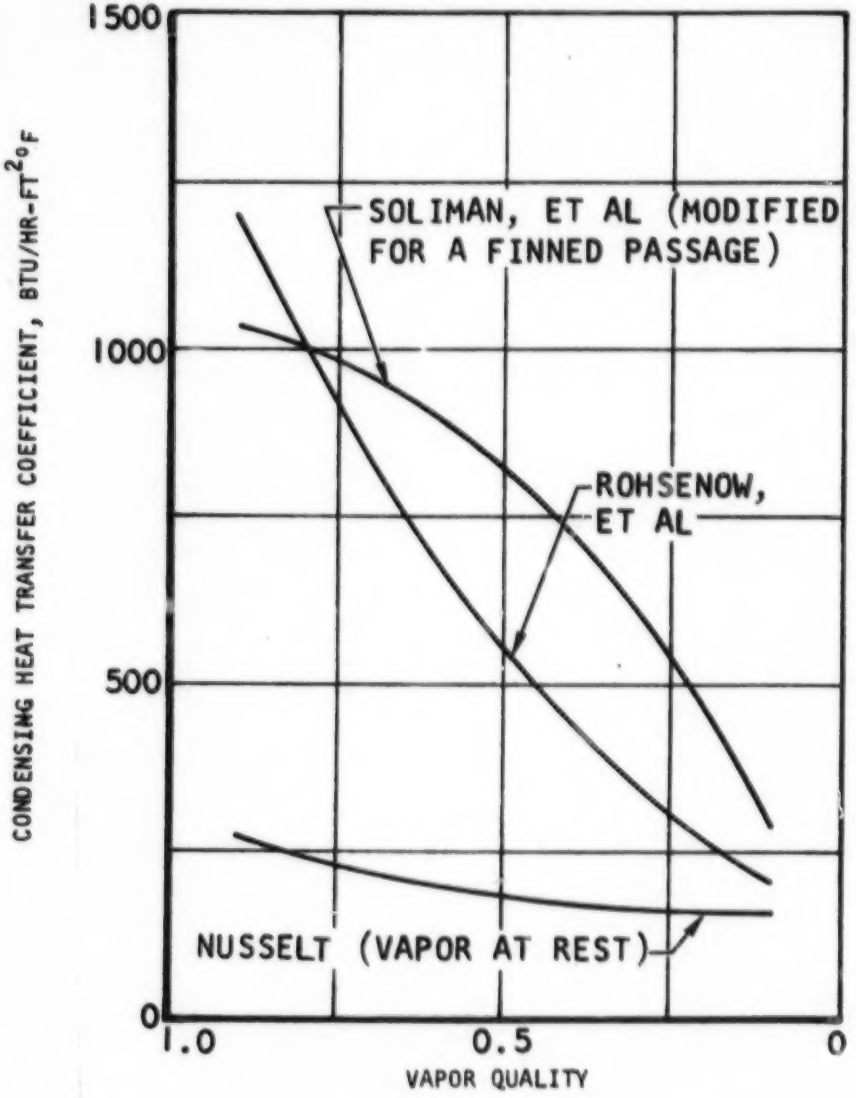


Figure 3. Perforated Fin Test Setup Schematic



CONDENSER DESIGN AND OPERATING CONDITIONS ARE THOSE OF THERMO ELECTRON CORPORATION

WORKING FLUID = FLUORINOL - 85, FLOWING VERTICALLY DOWNWARDS

S-65881

Figure 4. Local Condensing Heat Transfer Coefficients

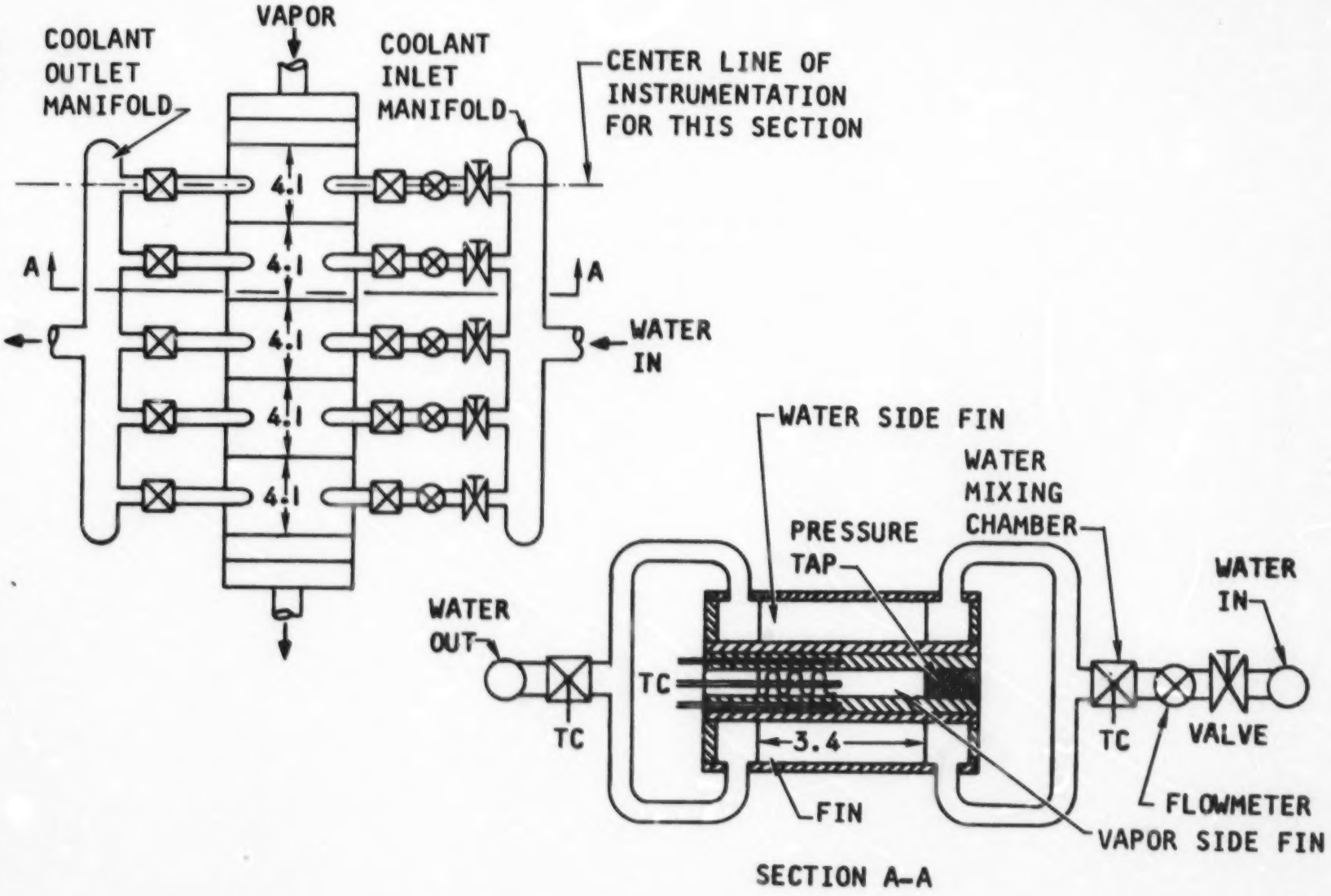


Figure 5. Schematic of Condensing Coefficient Test Unit

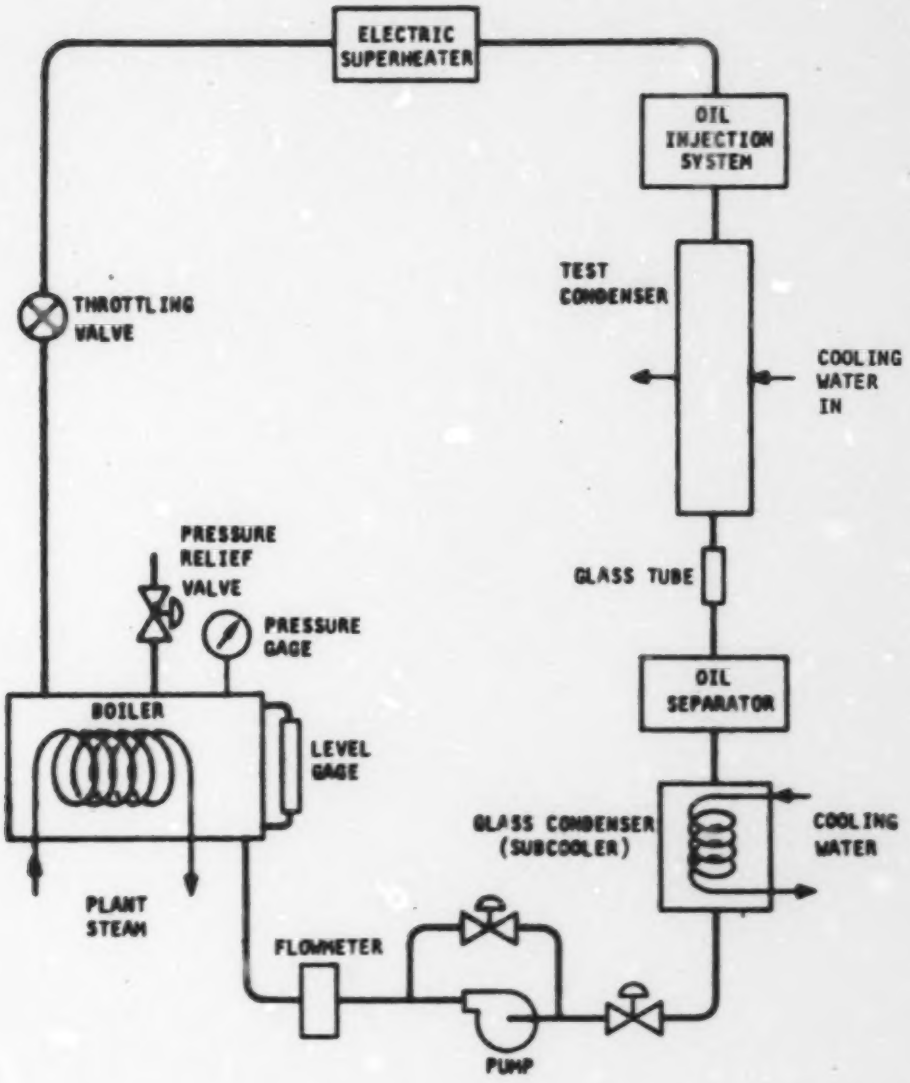


Figure 6. Single Channel Condensing Test Facility

TASK 3 -- DESIGN AND FABRICATE CONDENSERS

Preliminary condenser sizes were calculated for each of the three systems (TECO, Aerojet, and SES). A comparison of problem statements for the three condensers, as supplied to AiResearch by the individual contractors, is shown in Table I. It may be seen that the only condition common to all three condensers is an inlet air temperature of 85°F. In addition to the variation in problem statements generated by the differing system approaches to the Rankine-cycle, each condenser is designed for installation in a different vehicle and thus has different limitations on frontal area and heat exchanger depth. The design approach taken here is to utilize the maximum available frontal area for each condenser and size the airflow length to meet the particular problem statement. Thus the airflow lengths and the air pressure drops presented here are calculated values rather than design specifications. It should also be realized that airflow length may have a secondary effect on overall system performance in that it interrelates with the packaging arrangements of other engine components to establish inlet and exit losses and affects airflow distribution. These effects are not reflected in the comparisons made here and require additional analysis of the overall fan/condenser systems.

A comparison of condenser designs is given in Table 2. All three use the -13 perforated fin (cf. AiResearch Report 71-7464) on the air side, since this fin has been shown in previous studies to have the best f and j characteristics of the 11 perforated fin matrices tested by AiResearch. The -13 fin has a fin pitch of 22 fins per inch, a fin height of 0.326 in., and a fin thickness of 0.004 in. The perforations are rectangular slots with a slot width of 0.030 in. and a slot spacing such that a 25 percent open area is obtained. Other features of the condenser designs are discussed below for each condenser. Air pressure drops listed in Table 2 include core loss, a one-half velocity head loss at the core entrance, and an exit loss based on core attachment to a duct with a flow area equal to the frontal area of the core.

TECO Condenser

The TECO condenser is shown pictorially in Figure 7. The Fluorinol 85 working fluid enters the condenser at 40 psia and 238°F, condenses in the range of 217°F to 210°F (corresponding to 5 psi pressure drop), and leaves at 193°F. The condenser is thus designed for approximately 21°F de-superheat and 17°F subcooling in addition to the condensing load. Total heat rejection is 1.88×10^{-6} Btu/hr. TECO specification of the condenser problem statement called for an air-side effectiveness of 80 percent, which was applied to the average condensing temperature of 213°F to yield an airflow rate of 76,600 lb/hr. An effectiveness of 80 percent represented a reasonable initial estimate of optimum airflow on the basis of previous studies. These studies (cf 71-7464) showed (1) frontal area minimized at an effectivenss of about 0.86 for the preliminary problem statement (constant heat rejection and a fan power of six hp per million Btu/hr), and (2) weight decreased with decreasing effectiveness below this value Since the available core frontal area in the TECO problem statement (8.4 ft^2) is somewhat greater than the minimum frontal area obtained

TABLE I

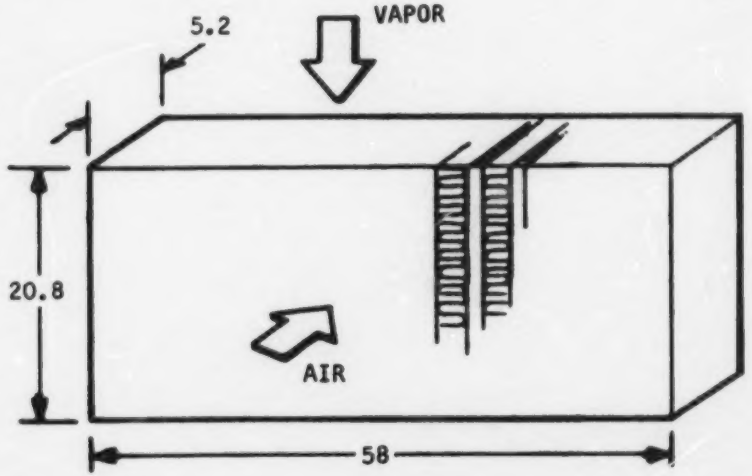
COMPARISON OF CONDENSER PROBLEM STATEMENTS

	TECO	SES	AEROJET
Working Fluid	Fluorinol 85	Steam	AEF-78
Total Heat Rejection (Btu/hr)	1.88×10^6	1.27×10^6	1.58×10^6
Condensing Temperature (^{0}F)	213	258	230
Subcooling (^{0}F)	17	None	60
Air Inlet Temperature (^{0}F)	85	85	85
Air-Side Effectiveness	0.8	0.8	0.63 (Condenser) 0.52 (Subcooler)
Vapor Flow Rate (lb/hr)	9,860	1,280	19,300

TABLE 2

COMPARISON OF CONDENSER DESIGNS

	TECO	SES	AEROJET
Core Frontal Area (ft^2)	8.46	6.46	6.05
Core Dry Weight (lb)	129	78	76
Air Flow Rate (lb/hr)	76,600	38,200	75,000
Air Pressure Drop (in. H_2O)	4.77	2.02	6.50
Air Horsepower	15.6	3.5	17.7
Vapor Pressure Drop (psi)	3	1	0.5



AIR FIN	22R-0.326-PERF(-13)-0.004(Al)*
VAPOR FIN	20R-0.050-OFFSET-0.004(Al)*
HEAT EXCHANGER CORE FRONTAL AREA	8.46 FT^2
HEAT EXCHANGER CORE WEIGHT	129 LB
AIR FLOW RATE	76,600 LB/HR
AIR PRESSURE DROP	4.77 IN. H_2O
VAPOR PRESSURE DROP	3 PSI
TOTAL HEAT REJECTION	1.88 X 10^6 BTU/HR

*Fin geometry listed in following order:
Rectangular fins per in. - fin height - fin type - fin thickness (material)

S-68083

Figure 7. Thermo Electron Condenser

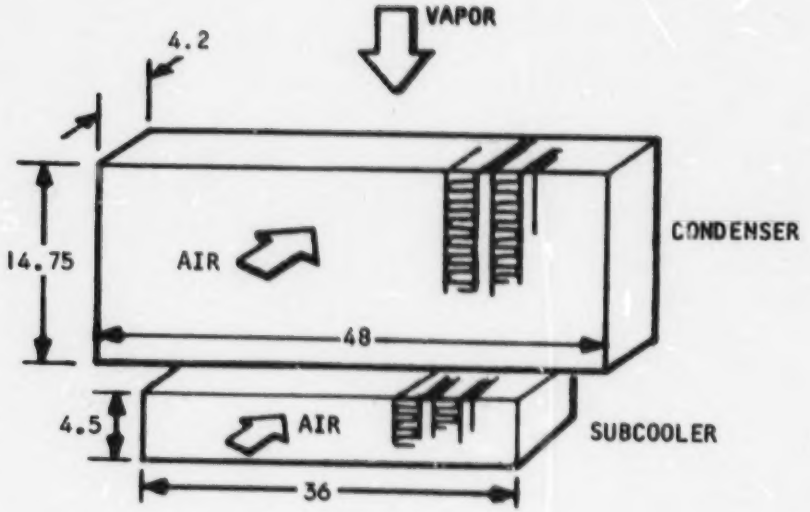
for the preliminary problem statement (in the range of 3.5 to 4.0 ft^2 per million Btu/hr), it can be expected that minimum fan power for the TECO design conditions will occur at a somewhat higher effectiveness than was obtained for the preliminary conditions. Considering the high fan power calculated for the present design at an effectivenss of 80 percent, it should be worthwhile to examine the effect on condenser design and fan power of reducing airflow from the current value of 76,600 lb/hr. Although such a reduction in airflow reduces fan power, it increases condenser airflow length, which already exceeds by 0.2 in. the maximum length of 5.0 in. specified by TECO. It thus appears that a relaxation of this maximum airflow length limitation could be beneficial to overall fan/condenser performance.

The total frontal dimensions of the condenser are 58 by 22.5 in., as per TECO specification, leaving 1.7 in. in the vapor-flow dimension for inlet and outlet manifolds. The vapor-side fin passages utilize an offset fin with a height of 0.050 in. and a pitch of 20 fins per inch. Core pressure drop for the vapor is estimated to be 3 psi. Since this is less than the maximum design value of 5 psi it may be possible to increase fin density on the vapor side (e.g., to 24 or 28 fins per inch) and effect a slight reduction in heat exchanger volume. Since the heat transfer conductance ratio for this heat exchanger design is approximately 3/1 with the air side controlling, each four percent increase in vapor-side conductance reduces heat exchanger volume by about one percent. A more detailed analysis of vapor pressure drop, including inlet and outlet manifold losses, is required to determine whether vapor fin density can be increased, and such an analysis will be performed as part of the detailed final design.

Calculation of the vapor-side conductance includes provision for the presence of lubricating oil, which enters the vapor stream in the reciprocating engine. The amount of oil entering the vapor stream, and the degree to which it collects in the condenser, are unknown. For calculation purposes, it was assumed that there would be a 0.001-in. thick oil film on all vapor-side heat transfer surfaces in the condenser. This assumption reduced condensing-side conductance by 43 percent and overall exchanger UA by 11 percent. The validity of this assumption will be determined in the condensing coefficient tests with and without oil injection in Task 2.

Aerojet Condenser

The Aerojet condenser consists of two separate heat exchangers, arranged in a T-shaped configuration, as shown in Figure 8. The sizes of the two units are such that essentially all of the condensation plus a small amount of subcooling occur in the upper unit, with the lower unit providing further subcooling. An intermediate manifold is required to collect and redistribute the fluid flow to the lower unit. The AEF-78 working fluid enters the condenser as a saturated vapor at 28.5 psia and 230°F and leaves as a subcooled liquid at 170°F. The problem statement thus requires essentially no de-superheat and approximately 60°F of subcooling. Total heat rejection is 1.58×10^6 Btu/hr, of which 0.28×10^6 Btu/hr is associated with subcooling.



AIR FIN	22R-0.326-PERF(-13)-0.004(AL)*
VAPOR FIN (CONDENSER)	12R-0.075-PLAIN-0.004(AL)*
VAPOR FIN (SUBCOOLER)	20R-0.075-OFFSET-0.004(AL)*
HEAT EXCHANGER CORE FRONTAL AREA	6.05 FT^2
HEAT EXCHANGER CORE WEIGHT	76 LB
AIR FLOW RATE	75,000 LB/HR
AIR PRESSURE DROP	6.50 IN. H_2O
VAPOR PRESSURE DROP	0.5 PSI
TOTAL HEAT REJECTION	1.58 X 10^6 BTU/HR

*Fin geometry listed in following order:
Rectangular fins per in. - fin height - fin type - fin thickness (material)

S-68082

Figure 8. Aerojet Condenser

Total airflow, as specified by Aerojet, is 75,000 lb/hr. Since the two units utilize the same air-side geometry and flow length, the total flow splits in proportion to the frontal areas, with 61,000 lb/hr going to the condenser and 14,000 lb/hr to the subcooler. Resultant effectivenesses for the two units (based on equal NTU's) are 63 percent for the condenser and 52 percent for the subcooler. These effectiveness values are much lower than previous studies have shown to be optimum (80 to 90 percent) and result in the relatively high air horsepower as shown in Table 2. Airflow for this condenser will be varied in future studies aimed at reducing fan power. These studies will include (1) the effect of losses external to the condenser plus the effect of ram air recovery in determining required fan performance as a function of airflow and (2) the trade-off between condenser size and fan power.

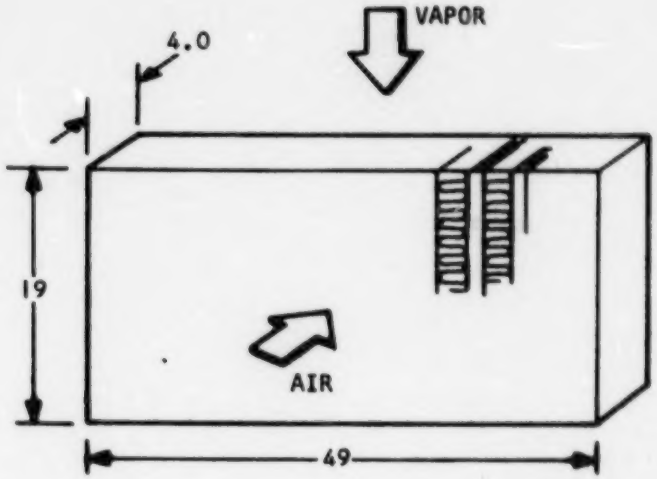
Core frontal dimensions are 14.75 in. by 48 in. for the condensing unit and 4.5 in. by 36 in. for the subcooler. Addition of inlet and outlet manifolds to both cores increases the height dimensions to about 16.75 in. for the condenser and 5.25 in. for the subcooler, for a total height of 22 in.

The condensing pressure drop of 0.5 psi is somewhat less than that allowed The use of offset fins in the condensing section was found to result in excessive pressure drop. Future design refinements should include increasing the number of fins per inch on the vapor side to increase vapor conductance and/or decreasing vapor-side fin height to provide more air-side passages. The heat transfer conductance ratio for the design of Figure 8 is 2.8 to 1 in the condenser and 2.2 to 1 in the subcooler, with the air side controlling in both units.

SES Condenser

The SES condenser is shown in Figure 9. The working fluid (steam) condenses at 258°F and 34 psia, rejecting 1.27×10^6 Btu/hr. The SES problem statement does not call for de-superheat or subcooling of the working fluid. Air-side effectiveness is 80 percent. As in the cases of the other two condensers, minimum fan power for the SES design probably occurs at a somewhat higher effectiveness than 80 percent. The situation with SES is somewhat different, however, in that the air horsepower is already relatively low (3.5 hp for condenser alone), so there is less to gain by reducing air flow. It should also be noted that the airflow length of the condenser is already somewhat greater than the 3.4 in. called for by SES and this would increase with increasing effectiveness (decreasing airflow). Further studies should include the trade-off between condenser size and fan horsepower in determining optimum effectiveness.

Core frontal dimensions are 19 in. by 49 in., as specified by SES. Use of an offset fin with a height of 0.050 in. and a pitch of 20 fins per inch on the vapor side yields a vapor pressure drop of about one psi. The use of a higher fin density on the vapor side (e.g., 24 or 28 fins per inch) is probably not warranted here because of the already high ratio of steam-side conductance to air-side conductance (five to one for the design of Figure 9). An investment of higher vapor pressure drop could more profitably be used by increasing the air-side fin height, thus reducing the number of vapor passages.



AIR FIN	22R-0.326-PERF(-13)-0.004(AL)*
VAPOR FIN	20R-0.050-OFFSET-0.004(AL)*
HEAT EXCHANGER CORE FRONTAL AREA	6.46 FT^2
HEAT EXCHANGER CORE WEIGHT	78 LB
AIR FLOW RATE	38,200 LB/HR
AIR PRESSURE DROP	2.02 IN. H_2O
VAPOR PRESSURE DROP	1 PSI
TOTAL HEAT REJECTION	1.27 X 10^6 BTU/HR

*Fin geometry listed in following order:
Rectangular fins per in. - fin height - fin type - fin thickness (material)

5-68084

Figure 9. Steam Engine Systems Condenser

The effect due to the presence of lubricating oil in the vapor stream was calculated, using the same assumption that was made for the TECO condenser, i.e., a 0.001-in.-thick oil film on all vapor-side heat transfer surfaces. This film reduced condensing-side conductance by 56 percent and overall exchanger UA by 10 percent. As in the case of the TECO condenser, the validity of this assumption can best be evaluated from condenser heat transfer tests with and without oil injection.

The conceptual design review has been held with TECO and the SES and Aerojet reviews are scheduled for early March.

An existing heat exchanger performance computer program is being modified to calculate condenser performance. This program will be used in the trade-off studies and in the final design of the condensers.

TASK 4 -- DETERMINE OPTIMUM COOLING SYSTEM INSTALLATION

The study of the effects of the automobile grill, air conditioning condenser, and engine compartment on the system airflow distribution and pressure losses was begun. The systems contractors were contacted to determine how they evaluated the airflow pressure losses and what the magnitude of these losses was. TECO evaluated the losses by assigning loss coefficients to the various elements in the system, SES assumed the losses were equal to three condenser exit velocity heads, and Aerojet assumed the total losses were equivalent to 25 percent of the ram air velocity head. The system pressure losses (other than condenser pressure drops) were all about 1 or 2 in. H_2O.

Test data of the pressure losses through the grill, air conditioning condenser, radiator, fan shroud, and engine compartment of a 1971 Galaxie as a function of air throughflow was obtained from Ford. The data was obtained in a wind tunnel, and, as indicated by Ford, does not correlate accurately with driving conditions. However, it should give an idea of the magnitude of the actual losses. Using loss coefficients calculated from the Ford data, the pressure drops through the Galaxie grill, the TECO condenser, and the Galaxie engine compartment were calculated at the TECO design airflow rate to be 3.60, 3.50, and 8.79 in. H_2O, respectively. In order to reduce the grill pressure loss the grill area will have to be increased. (This would be necessary anyway to achieve good condenser flow distribution.) It is not known how accurately the losses through the engine compartment with an IC engine represent the losses through the TECO rankine cycle engine compartment. If they are close, modifications must be made to reduce the losses or the fan input power will be unreasonably large. In order to determine the magnitude of the actual engine compartment losses as well as other system losses, mockup flow tests will be performed as discussed in Task 5 below.

TASK 5 -- DESIGN AND FABRICATE FANS

During the first quarterly reporting period, two tentative designs were developed to fulfill the condenser airflow requirements for the TECO engine. The first design assumed higher pressure drops (4.1 in. H_2O at design point conditions) as estimated by AiResearch. A second design was generated as

requested by TECO, for the same flow rating but only 3.1 in. H_2O total pressure rise across the fan.

This new problem statement is as follows:

Total flow through 2 fans:	75,300 lb/hr
Temperature at inlet to fans:	191°F
Total pressure rise across fans:	3.1 in. H_2O
Minimum total efficiency:	70 percent
Total volume flow:	20,600 cfm

The preliminary design for this problem statement is shown in Figure 10. It has been computer-calculated assuming evenly distributed flow across the fan. The final design will be made based on the actual flow distribution which will be measured on the installation model of the fan and condenser presently being constructed.

Table 3 gives the pertinent data of the fan design.

The design philosophy follows the usual procedure for the design of axial compressors with one exception: the flow pattern is assumed to be conical. This arrangement has two advantages: (1) the flow path of the air can clear the obstructions immediately behind the fan and (2) since the fan operates without stators the conically directed flow can recover more of the swirl than the pure axial one, thereby producing higher static pressures.

Figure 11 shows the calculated pressure rise, power requirement and efficiency of the fan as a function of the volume flow. The values are corrected for 14.7 psia pressure and 520°R temperature inlet condition. To obtain the actual pressure rise the following equation is valid:

$$\Delta P_{actual} = \Delta P_{diagram} \times 13.1 \times \rho_{inlet}$$

where ρ_{inlet} is the inlet air density in lb/cu ft.

The power is calculated similarly:

$$HP_{actual} = HP_{diagram} \times 13.1 \times \rho_{inlet}.$$

The curves are given for 2500 rpm speed.

To be able to estimate the noise output of the fan, a statistical diagram has been prepared showing the band of sound pressure output in decibels as a function of a parameter in Figure 12. This parameter is:

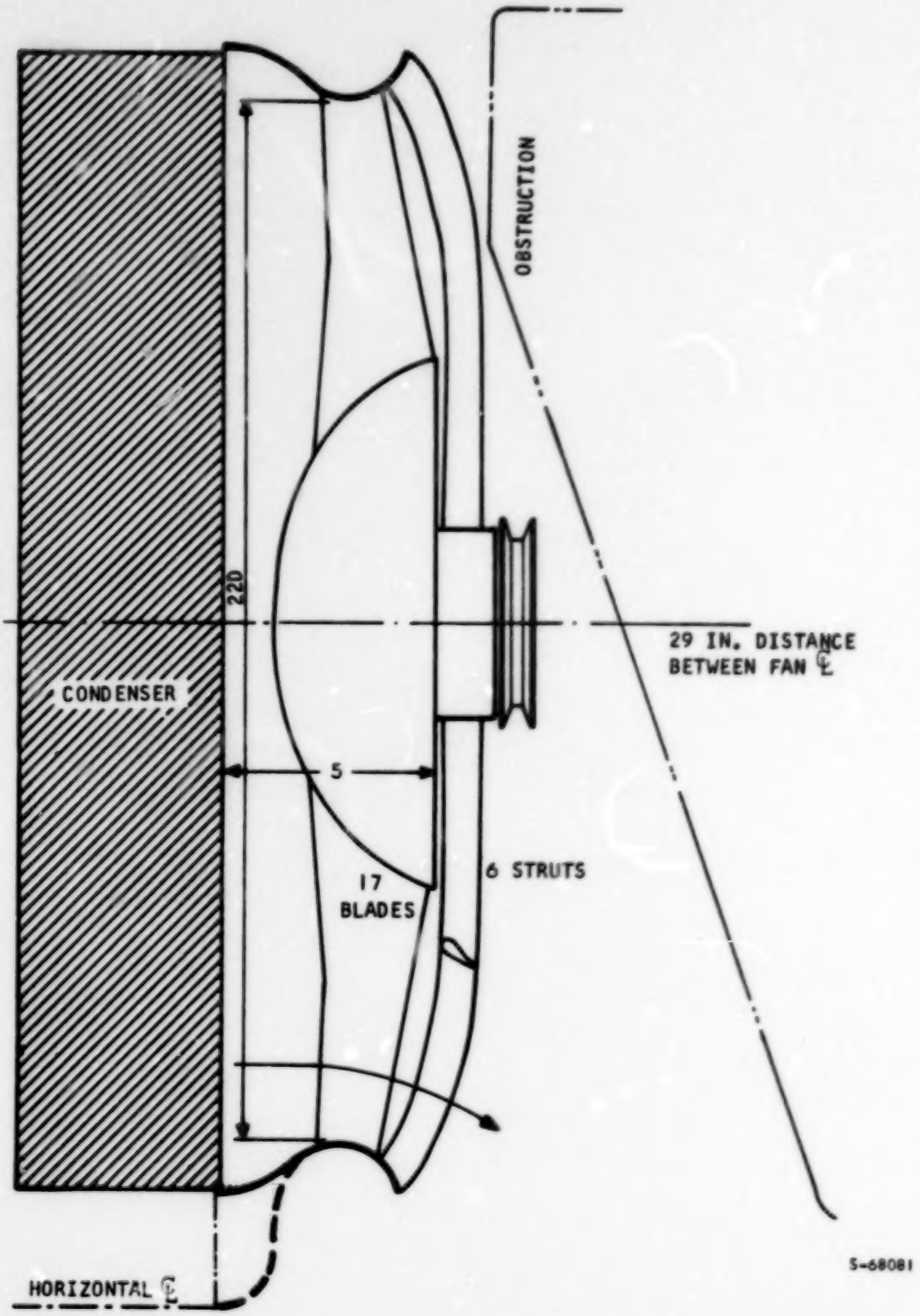


Figure 10. Fan Design

TABLE 3

FAN FOR RANKINE CYCLE CONDENSER
(PRELIMINARY DESIGN FOR TECO PROBLEM STATEMENT)

Total Pressure Rise	3.1 in. H_2O
Euler Pressure Rise	3.808 in. H_2O
Stage Efficiency	86 percent
Total Efficiency	70 percent
Design Flow	10,300 cfm 6275 lb/min
Shaft Power	7.2 hp
Speed	2500 rpm
Tip Diameter	22 in.
Hub Diameter (In)	9.4 in.
Hub Diameter (Out)	11.0 in.
Number of Blades	17
Profiles	NACA 65 Series
Tip Chord Length	2.950 in.
Hub Chord Length	3.080 in.
Tip Blade Angle (In)	70.43 deg
Tip Blade Angle (Out)	66.47 deg
Hub Blade Angle (In)	57.10 deg
Hub Blade Angle (Out)	1.014 deg

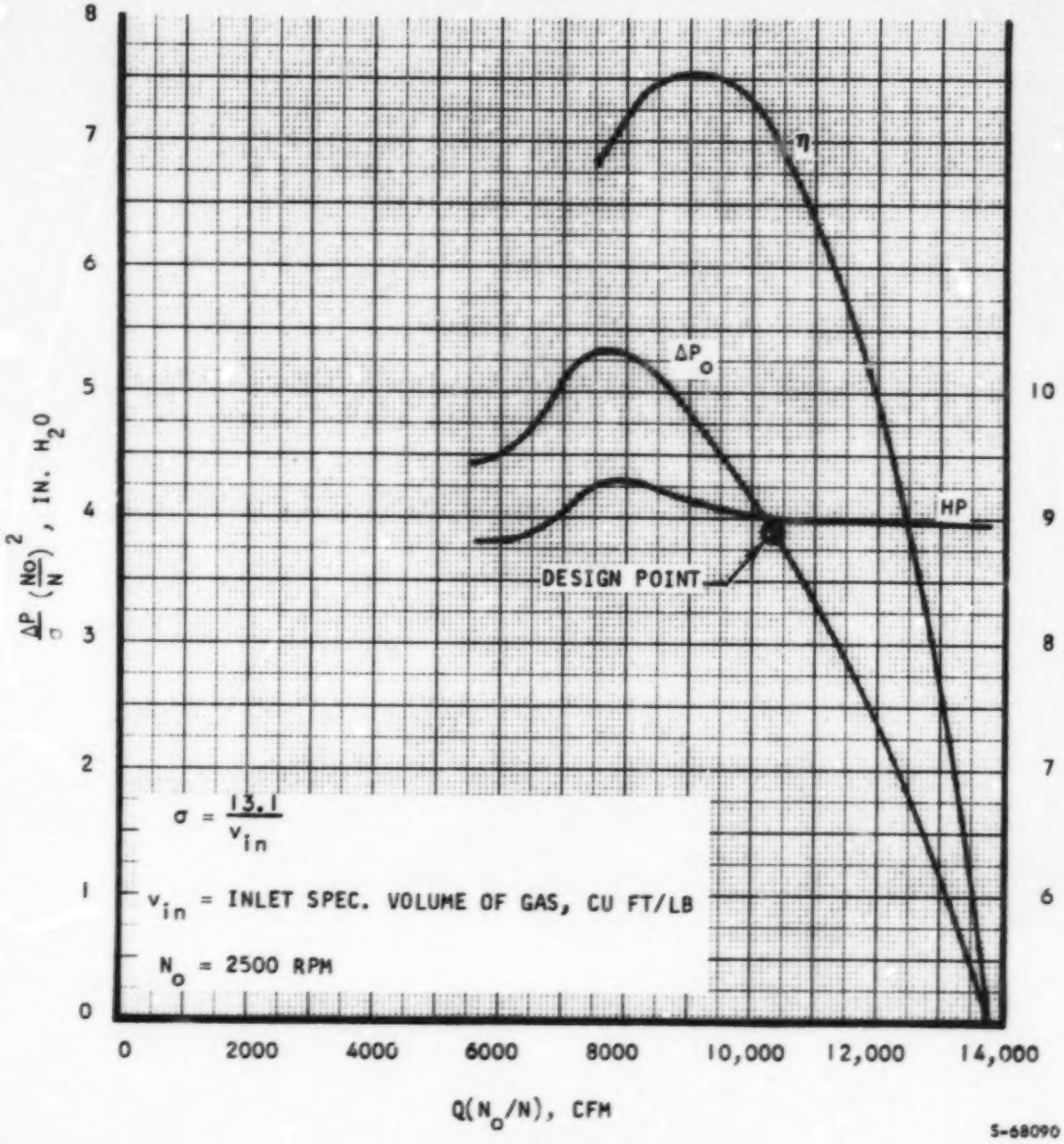


Figure 11. 3.1 in. H_2O Fan for Rankine-Cycle Condenser (TECO Problem Statement)

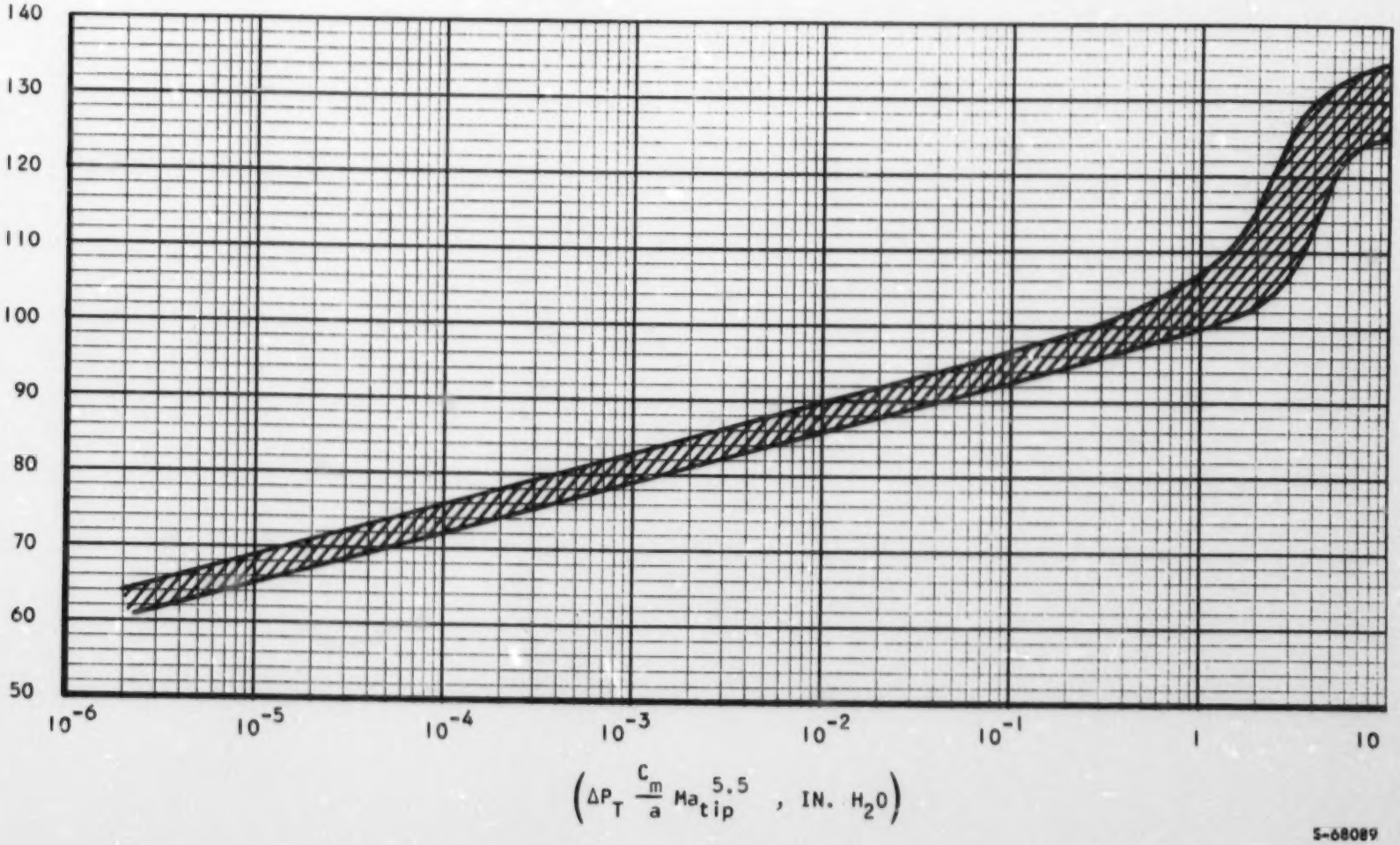


Figure 12. Statistical Sound Pressure Level of Fans

$$\Delta P_T = C_m/a \quad Ma_{tip}^{5.5}$$

where: ΔP_T = total pressure rise in inch H_2O

C_m/a = axial Mach no.

Ma_{tip} = tip Mach no.

The value of this parameter is 25×10^{-6} in the present case, therefore the expected noise pressure level is 72 decibels output from the fan itself. This value does not include interference noise which is generated due to uneven flow distribution or to the flow blowing across sharp edges or small holes.

As discussed under Task 4, data from Ford indicated higher system airflow losses than those anticipated by the system contractors. It appears desirable, therefore, to perform flow studies on Rankine engine mockups to determine the engine component pressure losses as well as the flow distribution which is critical to the fan design.

It was decided to procure a complete front end of a 1971 Ford Galaxie for the mockup installation. The burner, engine components, and other hardware likely to influence the fan discharge flow distribution were scaled off the available drawings and mockup parts are in fabrication. An artist's conception of the airflow mockup test is shown in Figure 13.

TASK 6 -- SYSTEM TECHNICAL COORDINATION

System contractor technical review meetings and the first contractor technical coordination meeting were held as scheduled.

TASK 7 -- DOCUMENTATION

The performance plan was issued in December and progress letters were prepared each month.

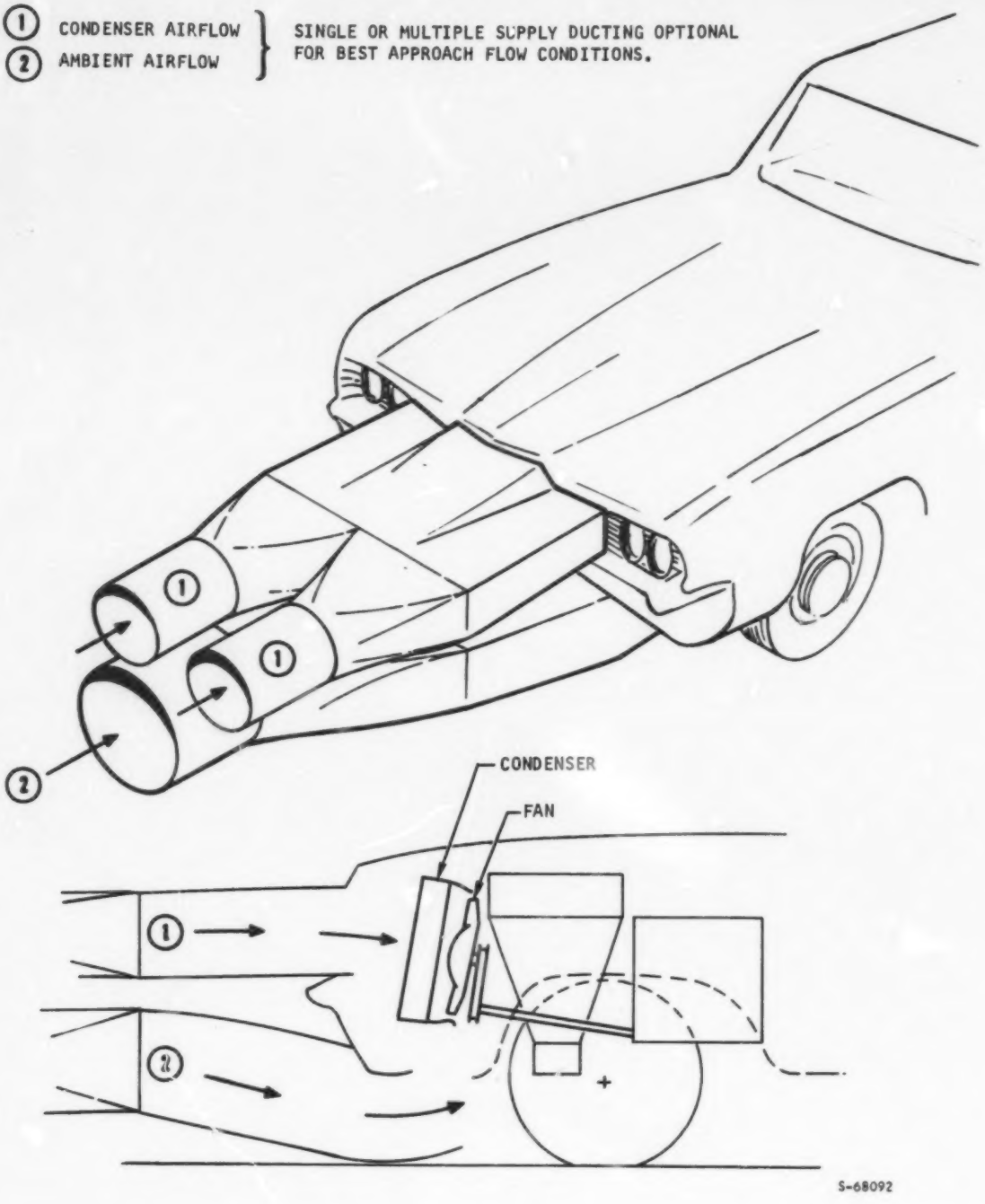


Figure 13. Rankine Cycle Mockup Test Setup—Flow Distribution in Typical Engine Compartment

REFERENCES

1. Soliman, M., J. R. Schuster, and P. J. Berenson, "A General Heat Transfer Correlation for Annular Flow Condensation," ASME Trans., Journal of Heat Transfer, 90, pp. 267-276, May 1968.

2. Rohsenow, W. M., D. P. Traviss, and A. B. Baron, Forced-Convection Condensation Inside Tubes, Report No. DSR 72591-74, Massachusetts Institute of Technology, Cambridge, Massachusetts, July 1971.

SECTION 3

SCHEDULE CHANGES

Schedule slippages have occurred in several areas. Mose of these slips are due to the compressed schedule which was the result of starting work in November while still maintaining the delivery dates which were based on an anticipated September 1 start date.

Selection of the third perforated fin geometry has slipped from December to February because the University of Michigan study upon which the selection of the third geometry was to be based indicated the optimum open area was covered in the scheduled tests. An analytical optimization of fin height will be performed to see if the third surface should be a different fin geometry. No new perforation tooling will be required and the fin will be made using an existing variable height die. Data reduction scheduled to begin in January will not begin until February because the cores will not be ready for testing until then. It is anticipated that these slips will delay the Task 1 completion date by two weeks.

The Task 2 schedule has been accelerated to finish earlier than originally scheduled so that the condensing coefficient data may be used in the full size condenser design.

The Aerojet and SES condenser conceptual design reviews have been rescheduled for early March.

Schedule changes for Tasks 3 and 5 are anticipated and have been submitted to the EPA for approval.

SECTION 4

CURRENT PROBLEMS

None

SECTION 5

PLANNED WORK

During the next quarter, the third perforated fin geometry will be selected, all three f and j test cores will be fabricated, and testing will be performed. The condensing test rig will be assembled, the test unit built, and the condensing heat transfer coefficient tests will begin. Conceptual design reviews will be held with Aerojet and SES. Tradeoff studies establishing the relationship between fan horsepower and condenser length vs functions of airflow rate and the fin geometry will be performed and final design of the condensers will begin. The engine mockup flow distribution tests will be performed and final designs of the fans begun.